TEKA Kom. Mot. Energ. Roln., 2006, 6, 102–108

INCREASES OF NOMINAL PRESSURE AT HYDRAULIC UNLOADING OF THE SWINGING UNIT IN THE AXIAL-PISTON HYDROMACHINE

Ilya Nikolenko*, Eugeniusz Krasowski**, Victor Nikolenko***

*Odessa Agricultural University

**Agricultural University of Lublin

***Odessa National Polytechnic University

Summary. An establishment of dependences for a definition of nominal pressure of the axial piston hydromachine, whose fatigue is limited by durability of the block of cylinders, is presented. Results of calculation of intensity of pressure in the crosspiece between cylinders are given, depending on the basic geometrical parameters of the swinging unit. An analysis of influence of the length of cylinders, forms of a sealing part of pistons and rigidity of a ground part of the block of cylinders on the level of nominal pressure of the hydromachine is carried out.

Key words: nominal pressure, hydraulic unloading

The development of new axial piston hydromachines (APH), and their modernization begins with the development of swinging units (SU) that is inextricably related with a rational choice of the constructive sizes of cylinder piston group (CPG). The complexity of such choice causes that in the development of SU it is necessary to take into account various, at times inconsistent, requirements. Sizes CPG should provide given working volume and efficiency of the hydromachine, durability, rigidity, wear resistance and durability of details. Thus sizes SU determine the dimensions, weight and the moment of inertia of the hydromachine.

The initial data at designing SU APH are a working volume of the hydromachine – the maximal and nominal pressure in a hydroline, durability, tribotechnical characteristics of the material, the block of cylinders (BC) and details of piston group, durability and efficiency of the hydromachine. As a result of design calculation it is necessary to define basic geometrical parameters SU: number and radius of cylinders, radius of an arrangement of their centres, a corner of an inclination. The traditional approach at design calculations of geometrical parameters BC is their definition from a condition maintenance of durability from fragile destruction at action of the maximal pressure.

Change of a technical condition of units of hydrodrives under operating conditions occurs as a result of wear process and fatigue failures of basic details SU. Consequences of change of a technical condition of units of hydrodrives, and also losses of their serviceability under an influence of these factors are not identical. The wear process results in gradual refusals which are shown by a decrease of efficiency of units, an increase of vibrations and the level of noise. Depending on many factors, such as: kinds of interfaces of details, modes of operation of hydrounits, temperature conditions, quality of the working liquid etc., the surface of details can be exposed to various kinds of wear process: abrasive, hydroabrasive, erosive, oxidizing, and also contact weariness.

The most effective ways of increasing the nominal pressure APH are an application of polymetallic and compound designs of details SU which allow to apply the principle of local quality. These various parts and surfaces of details are provided with specific properties. One of such ways in volumetric hydromachines is hydraulic unloading of details. Hydraulic unloading CPG – increase of durability and rigidity BC by the appendix on its external surface of efforts of pressure of a working liquid which are connected to pressure in a line of high pressure APH and with position of pistons in cylinders.

Hydraulic unloading allows to apply for manufacturing of BC the antifriction materials with rather low durability characteristics, but good antifriction characteristics which provide SU high hydro-erosion resistance, wear resistance and reduce mechanical losses of capacity. A doubtless advantage of hydraulic unloading is stabilization of the form and sizes of a backlash in CPG at an increase of working pressure that allows to increase efficiency and durability of the hydromachine. Absence of methods of definition of nominal pressure APH on a design stage does not allow to execute an estimation of various variants of designs, and demands experimental check of many variants, that finally raises cost of machines and increases terms of their manufacturing.

Fatigue failures of SU details are more dangerous because, as a rule, they result in sudden refusals which can cause occurrence of emergencies at damage or breakages of other details. Restoration of serviceability APH at a sudden SU refusal is connected to significant costs which are caused by necessity of dismantle of hydromachines and their major overhaul. The data of refusals APH, the analysis of the character of destructions, showed, that sudden failures of a base detail of APH – BC – carry basically a fatigue character [Nikolenko *et al.* 1988].

The research accounts for a cyclic operation of pressure in cylinders of the block, non-uniform distribution of stress and deformations in their axial and radial sections, and also presence of stress concentrators in BC. In this connection, in design calculations of CPG it is also necessary to satisfy the nominal pressure from a condition of performance durability BC. A design procedure BC on durability is offered in works [Nikolenko 2001a, b] according to which the settlement circuit BC in spatial statement represents a system of elastically connected beams with the rigid centre. Given the settlement circuit BC, it is possible to investigate its stress - deformed a condition (IDC) with the purpose of definition of normal pressure in its dangerous zones and finding in them the dangerous points where the maximal cyclic pressure operates. For the plastic materials used for manufacturing BC, in case of the complex IDC equivalent pressure, it is defined under the fourth theory of durability by the criterion of specific potential energy change of the form, the Hyber-Henky criterion. The equivalent pressure under the fourth theory of durability is named intensity of stress.

Accumulation of fatigue damages is connected to the course of cyclic plastic deformations and conditions of durability at variable loadings and the complex intense condition have similar conditions on criteria of durability with replacement of stress by their peak values. Therefore for maintenance of durability BC at base number of cycles in a dangerous point of section it is necessary to satisfy the condition:

$$\sigma_{i\max} = p_{nom} \cdot \overline{\sigma}_{i\max} \le \sigma_0 \tag{1}$$

where:

$\sigma_{i\max}$ – peak value intensity of stress in dangerous point BC,

p_{nom} – nominal pressure APH,

$\overline{\sigma}_{i\max}$ – factor of intensity of stress in a dangerous point,

σ_0 – fatigue strength of material BC at about zero cycle of loading .

In view of the factor intensity of stress in dangerous zones BC, the value of nominal pressure APH from a condition of durability (1) is defined on the dependence:

$$p_{nom} \le \frac{\sigma_0}{\overline{\sigma}_{i\max}} \tag{2}$$

For dangerous zones BC the factor intensity of stress according to [Nikolenko 2001a, b] is represented as:

$$\overline{\sigma}_i(\xi,\xi_0) = \sqrt{0{,}5 \cdot \left[\overline{\sigma}_{xy}^2(\xi,\xi_0) + \overline{\sigma}_{xz}^2(\xi,\xi_0) + \overline{\sigma}_{yz}^2(\xi,\xi_0)\right]} \tag{3}$$

where:

$\xi = ax$, $\xi_0 = ax_0$ – dimensionless coordinates of section BC and a bottom of the piston,

$a = \sqrt[4]{\overline{K}/4 \cdot I_0}$ – relative parameter BC which is defined by its radial sizes and elastic characteristics of a material,

$\overline{K}$ – factor of relative rigidity of the basis,

x, x_0 – axial coordinates of considered section BC and a bottom of the piston in one of cylinders,

$\overline{\sigma}_{xy}(\xi,\xi_0) = \left[\overline{\sigma}_x(\xi,\xi_0) - \overline{\sigma}_y(\xi,\xi_0)\right]$, $\overline{\sigma}_{xz}(\xi,\xi_0) = \left[\overline{\sigma}_x(\xi,\xi_0) - \overline{\sigma}_z(\xi,\xi_0)\right]$,

$\overline{\sigma}_{yz}(\xi,\xi_0) = \left[\overline{\sigma}_y(\xi,\xi_0) - \overline{\sigma}_z(\xi,\xi_0)\right]$, and $\overline{\sigma}_x(\xi,\xi_0)$, $\overline{\sigma}_y(\xi,\xi_0)$, $\overline{\sigma}_z(\xi,\xi_0)$ –

– factors of concentration of stress of the appropriate direction.

For BC the most dangerous zones from a condition of its durability are crosspieces between cylinders in places of their minimal thickness. [Nikolenko 2001a] The factors of stress concentration received in work in the crosspiece between cylinders we shall present as:

Axial in the crosspiece

$$\overline{\sigma}_x(\xi,\xi_0) = \beta \cdot \overline{M}(\xi,\xi_0) \tag{4}$$

where:

$\beta = \dfrac{2 \cdot \overline{a}^2 \cdot \left(\overline{y}_0 - \left(R/r_ц\right) \cdot \cos\left(\pi/z_ц\right)\right) \cdot \left(1 + \sin\left(\pi/z_ц\right)\right)}{\overline{K}}$ – factor of the moment of resistance of section on a bend,

$\overline{y}_0$ – relative coordinate the centre of gravity of section,

R – radius of accommodation of the centres of cylinders,

$r_ц$ and $z_ц$ – radius and number of cylinders,

$\overline{M}\left(\xi, \xi_0\right)$ – the relative bending moment;

Radial in the crosspiece

$$\overline{\sigma}_y\left(\xi, \xi_0\right) = \alpha \cdot \overline{y}\left(\xi, \xi_0\right) \tag{5}$$

where:

$\alpha = \dfrac{r_ц \cdot \left(1 + \sin\left(\pi/z_ц\right)\right)}{B \cdot I_1\left(L_1\right) \cdot \overline{K}}$ – factor of rigidity of the crosspiece,

$B = \left(R \sin\left(\pi/z_ц\right) - r_ц\right)$ – thickness of the crosspiece between cylinders,

$I_1\left(L_1\right)$ – factor of a pliability of the crosspiece at action of stretching force,

$\overline{y}\left(\xi, \xi_0\right)$ – relative radial deformation of the crosspiece.

Except normal stress from a stretching of crosspieces and axial in sections BC from a bend, on a surface of cylinders, the pressure of a working liquid operates, which creates compressing pressure(voltage) in these zones. In the chosen system of coordinates, on the crosspiece between cylinders in a zone of its minimal thickness it is a district stress. On the length of the cylinder they correspond with the pressure working in the cylinder, in view of a designation of dimensionless length of a sealing part of the piston $m = aL_n$ look like

$$\overline{\sigma}_z\left(\xi, \xi_0\right) = \begin{cases} 0, & 0 \le \xi \le \xi_0 - m; \\ \dfrac{\xi_0 - \xi - m}{m}, & \xi_0 - m < \xi \le \xi_0; \\ -1, & \xi_0 < \xi \le \gamma. \end{cases} \tag{6}$$

Thus, the developed mathematical model allows to define stress and their intensity in dangerous zones BC. The maximal loading of crosspieces between cylinders arises at the maximal distance of the piston from a ground part of the cylinder, that is in its position in the top dead point with dimensionless coordinate

$$\xi_{0\min} = a \cdot \left[L_ц - l_0 - 4R \cdot tg\left(\varphi/2\right)\right]$$

where:

l_0 – distance between a bottom of the piston and a bottom of the cylinder in a bottom dead point,

φ – a corner of inclination BC or a disk.

From the analysis of dependences (4) and (5) follows, that factors depend only on radial sizes BC α, β and $\overline{\sigma}_z(\xi, \xi_0)$, $\overline{M}(\xi, \xi_0)$ and $\overline{y}(\xi, \xi_0)$ depend on its axial sizes and kinematics APH.

In view of values of factors of stress concentration (4) – (6) the maximal factor of intensity of stress we shall present as

$$\overline{\sigma}_i(\xi, \xi_{0\min}) = \sqrt{0{,}5 \cdot \left[(\alpha y \mp \beta M)^2 + (1 \pm \beta M)^2 + (\alpha y + 1)^2\right]} \tag{7}$$

Here it is designated $M = \overline{M}(\xi, \xi_{0\min})$ and $y = \overline{y}(\xi, \xi_{0\min})$, and $\pm$ the mark M is applied to zones of a stretching and compression accordingly from action of the axial bending moment in the crosspiece.

Let's consider most dangerous zones BC which are junctions of crosspieces between cylinders with a ground part and crosspieces in a zone of the top dead point of one of pistons. According to the accepted settlement circuit, the section of the crosspiece is a zone of its connection with a ground part $\xi = \gamma$, $y = \overline{y}(\gamma, \xi_{0\min}) = 0$, also the moment which causes stretching stress. Having substituted these values in (9) we shall receive factor of stress intensity in the crosspiece between cylinders in their ground part

$$\overline{\sigma}_i(\gamma, \xi_{0\min}) = \sqrt{\beta^2 M^2 + \beta M + 1} \tag{8}$$

In section of the crosspiece in a zone of the top dead point $\xi_{0\min} = \gamma$, $y = \overline{y}(\gamma, \xi_{0\min}) \approx 1$, also the moment which causes pressure of compression operates. Results of calculation of relative radial deformation of cylinders are executed in work [Nikolenko and Krasowski 2002]. Accordingly, factor of stress intensity in the crosspiece

$$\overline{\sigma}_i(\xi_{0\min}, \xi_{0\min}) = \sqrt{0{,}5 \cdot \left[(\alpha + \beta M)^2 + (1 - \beta M)^2 + (\alpha + 1)^2\right]} \tag{9}$$

From the submitted dependences (8) and (9) follows, that the zone of the maximal stress intensity is determined by radial sizes BC, kinematics of the hydromachine and distribution of the relative bending moment in the crosspiece. For settlement estimation of influence of axial geometrical parameters BC and the form of a sealing part on the relative bending moment in the crosspiece we consider the simplified variant of an arrangement of pistons in the cylinders – accommodation of pistons in two next cylinders in a point with the coordinate. Calculations of the relative bending moments in the crosspiece between cylinders of the block for three values of relative rigidity of a ground part 0,1; 1,0 and 10,0 are executed. Pre-computations have shown, that at relative rigidity of 0.1 values of deformations correspond to indefinitely big rigidity of a ground part, that is 0, and values 10.0 correspond to zero rigidity, that is the influence of length of a sealing part of the piston was estimated by two limiting values of length of a sealing part of the piston end.

The analysis of the received results has also shown, that without dependence on the length of cylinders, relative rigidity of a ground part and length of a sealing part of the piston in zones with dimensionless coordinates $\xi = 0{,}5\gamma$ relative bending moments in the crosspiece between cylinders of the block $\overline{M}(\xi, \xi_0) \approx 0$. In view of this, the dependence (11) can be presented as:

$$\overline{\sigma}_i\left(0{,}5\gamma, \xi_{0\min}\right) = \sqrt{y^2\alpha^2 + y\alpha + 1} \qquad (10)$$

Hydraulic unloading of details and units is used in many designs of volumetric hydromachines with the purpose of increasing their working pressure and a resource. At hydraulic unloading, details on their surfaces cavities, whose channels incorporate to an operative range of working pressure, are executed. Total force of pressure of unloading on a wall of a cavity is directed to the opposite party to total force of pressure upon a detail or on its most loaded part. Therefore at hydraulic unloading, the unloading efforts are always proportional to the operating pressure, and consequently, to the operating loadings. An effective way of increasing the durability and rigidity of BC is to decrease the pressure in its dangerous points due to the appendix of pressure upon an external surface of the block.

The paper presents ways of hydraulic unloading of the swinging unit АПГ and the design procedure. In Fig.1 results of calculation by this technique of relative deformations BC for serial swinging unit of the hydromachine 300.25 at a corner of inclination BC 25^{o} are presented to submission of pressure from a cavity of cylinders and accommodation of cavities of hydro-unloading above external walls of cylinders. Relative deformations in three axial sections BC with coordinates are considered.

From the dependencies presented in Fig. 1 it is visible, that hydraulic unloading allows to lower pressure and deformations in BC by 2–4 times, which can provide an increase of the working pressure and efficiency of the hydromachine which are limited by durability and rigidity of the block.

The executed calculation has shown an opportunity to increase the durability and rigidity of the swinging unit at action of nominal pressure not less than twice, which allows to raise working pressure of the hydromachine without replacement of a material of the block by a material with higher durability characteristics.

The executed calculation has shown an opportunity to increase the durability and rigidity BC of serial swinging unit 300.25 at action of the maximal pressure not less than twice, which allows to raise (increase) working pressure of the hydromachine without replacement of a material of the block by a material with higher durability characteristics.

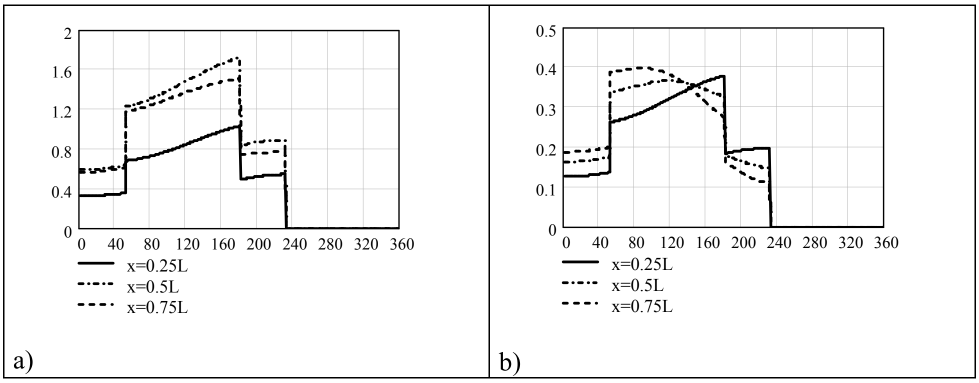

Fig. 1. Relative deformations $y = \bar{y}(\xi, \xi_0)$ of the swinging unit 310.25 from a corner of turn BC: a – BC without hydraulic unloading, b – BC with hydraulic unloading,

Thus, definition of zones of the maximal intensity of pressure in crosspieces between cylinders of the block at definition of a level of nominal pressure at design calculations APH consists in the definition of parameters on known radial sizes, and values and on known axial sizes SU in three sections BC and at position of the piston in the top dead point. As a result of a comparison of the received factors of intensity of pressure, dangerous zone BC on which, according to the dependence (2), the size of nominal pressure APH is established.

The submitted technique allows to take into account at a stage of design calculations the basic geometrical parameters BC at the definition of the nominal pressure APH values.

REFERENCES

Nikolenko I.V., Khomyak Yu.M, Kibakov A.G. 1988: Raschet na dolgovichnost bloka cylindrov hydromashiny. Visnik mashynostroyenia, 2, 26–29.

Nikolenko I.V. 2001a: Rosrakhunok deformacyi bloku cylindriv aksyalno-porshnievoy hydromashiny. Agrarniy vicnik Prichornomorya, 36 nauk. pr. Odeskiy DSGI, 4, 15, 135–143.

Nikolenko I.V. 2001b: Vliyanie chisla porshniey aksyalno-porshnievoy hydromashiny na parametry kachayushcheho uzla. Motorization and energetics in agriculture: III International research and technical conference. Lublin (Poland): WAR, 4, 281–286.

Nikolenko I.V., Krasowski E. 2002: Tendencies of development in the field of designs and calculation methods of axial piston hydromachines. Teka Kom. Mot. i Energ. Roln. 2, 149–157.